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A study on the starting characteristics of a reciprocating compressor for a household refrigerator

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Abstract

The starting characteristics of a reciprocating compressor used in household refrigerators are discussed in this paper.

Failure of starting the compressor, which usually occurs at low line voltage, causes the reduction of compressor lifetime. Therefore the starting characteristics in the compressor design must be carefully considered, especially where the line voltage is unstable.

Experimental investigations on the compressor starting voltage have been performed on the effect of design parameters such as discharge a valve, valve port area, piston pre-caulking torque with the compressor which has ball joint type connecting rod. Also numerical analysis has been carried out simultaneously in order to develop a computer simulation program.

Nomenclature

T_c	pre-caulking torque (assembling torque of ball and piston)
F_{bx}	x-directional force acted on the ball
F_{by}	y-directional force acted on the ball
$P(\xi)$	pressure distribution on the ball surface
r_b	ball radius
α	contact angle between ball and ball seat of piston
β	contact angle between ball and buffer ring
b_s	contact width of ball and ball seat of piston
b_t	contact width of ball and buffer ring
f_s	friction coefficients between ball and ball seat
f_t	friction coefficients between ball and buffer ring
I_{sh}	crank shaft moment of inertia.
ω	rotational speed of the crank shaft
t	operating time of starting relay
k	polytropic constant

Introduction

Not only the steady state performance but also the starting characteristics of a compressor are very important in the refrigerator compressor. The compressor motor is usually designed to have large torque to make compressor start smoothly, which reduces the steady state performance of the compressor.

Information about the starting characteristics of the compressor was mainly related with the starting torque of the motor to move the mechanical parts.^{[1][2]} In case that the refrigerating compressor fail to run under balanced pressure condition, the mechanical moving parts are accelerated within a few seconds. The failure in running is caused from the frictional resistance between sliding parts, the pressure resistance in cylinder volume and the accelerating time which is needed to reach steady state operating region.

In this paper, The effects of pressure condition in cylinder volume, frictional resistance of the mechanical parts and operating time of starting relay were investigated.

Experimental and calculational results show that pressure condition in the cylinder and friction resistance of ball joint part are very important in determining the starting characteristics of the compressor

Analysis

In order to calculate compressor load torque at each rotational speed, force and moment equilibrium equation of moving parts were used. the valve system was simplified as one dimensional mass-spring-damper system and the equation was solved using Runge-Kutta-Nystrom method.

The compressor motor torque should be equal to the sum of friction torque of mechanical parts, gas load torque and accelerating torque.

$$T_{\text{motor}} = T_f + T_g + T_a \quad \text{-----} \quad (1)$$

Friction part

1) Friction at the ball joint

The free body diagram of the piston and the ball is shown in Fig. 2. When F_{bx} is positive i.e. the piston is moving toward the valve, the friction occurs at the contact area between ball and ball seat of the piston. When F_{bx} is negative, the friction occurs between the ball and the buffer ring. In the former case, the friction coefficient f_s was used and in the latter case the friction coefficient f_t was used. It was assumed that pre-caulking torque T_c was constant, because the effect of direction and magnitude of F_{bx} , F_{by} on T_c was negligible.

In Fig. 2, the pressure distribution $P(\xi)$ on the ball surface can be written as follows

$$\int P(\xi) \sin(\alpha) dA = F_{by} \quad \text{-----} \quad (2)$$

$$\int P(\xi) \cos(\alpha) dA = F_{bx} \quad \text{-----} \quad (3)$$

Assuming that $P(\xi)$ is proportional to ξ , the pressure $P(\xi)$ can be expressed as Eq. (4)

$$P(\xi) = \frac{F_{by}}{4r_b b \sin^2(\alpha)} \cdot \xi + \frac{F_{bx}}{2\pi r_b b \cos(\alpha) \sin(\alpha)} - \frac{\pi F_{bx}}{8r_b b \sin^2(\alpha)} \quad \text{--} \quad (4)$$

The friction torque T_s , T_t caused by $P(\xi)$ are

$$T_s = f_s r_b \int (\cos^2(\alpha) + \sin^2(\alpha) \cos^2(\xi))^{1/2} P(\xi) dA_s \quad \text{-----} \quad (5)$$

$$T_t = f_s r_b \int (\cos^2(\beta) + \sin^2(\beta) \cos^2(\xi))^{1/2} P(\xi) dA_t \quad \text{-----} \quad (6)$$

where $dA_s = r_b \sin(\alpha) b_s d\xi$; $dA_t = r_b \sin(\beta) b_t d\xi$

2) Friction in the journal bearing

Below the rotational speed which can supply bearing parts with refrigeration oil, sliding friction coefficient was used. If the eccentricity of the shaft center in journal bearing is larger than 0.8, it is reasonable to assume that metal contact between each sliding surface occurs^[3].

The friction coefficient of hydrodynamic lubrication was obtained from the full journal bearing solution table by using Sommerfeld Number.

Cylinder pressure

It is assumed that gas flow through the discharge valve is one dimensional isentropic flow and that the compression process is treated as polytropic process model as the Eq. (7).

$$P_1 V_1^k = P_2 V_2^k \quad \text{-----} \quad (7)$$

Accelerating torque

From the Eq. (1), accelerating torque can be written by

$$T_a = T_{\text{motor}} - T_f - T_g \quad \text{-----} \quad (8)$$

The work done by the accelerating torque is converted to the kinetic energy of shaft and rotor and to the energy loss dissipated to heat. However the dissipation loss was neglected because the accelerating time is too short to consider the dissipation loss. Time t means accelerating time needed to reach steady running speed

$$\frac{1}{2} I_{sh} \omega^2 = \int T_a(\omega) d\omega \cdot t \quad \text{-----} \quad (9)$$

Experiment

Experiments were conducted with 1/4hp hermetic compressor used in household refrigerator. The load stand which could control the discharge and suction pressure automatically. HFC-134a was used as refrigerant. The pressure in cylinder volume was measured with small size pressure transducer. It is shown in Fig. 3. The rotational speed of the crank shaft was determined from the pressure-time trace on the data analyzer. The measurement error of the rotational speed was probably within ± 10 rpm^[4].

Friction coefficients of sliding parts were measured several times. Mean values(in table 1) were used in simulation program.

From the motor torque curve measured from 60 to 220 voltage at every 20 voltage, torque needed to maintain steady state condition at each rotating speed of the shaft was estimated. To measure the friction torque, compressor was tested under the vacuum condition and compressor rotational speed was measured with stroboscope through the transparent plastic window. Several piston assemblies which had different pre-caulking torque, were used in this experiment to verify the effects of the change of friction condition in the ball joint part.

To change gas pressure load, valve plates with different port sizes and discharge valves with different thickness were used. The electrical current that pass through the PTC type starting relay was monitored and the operating times of starting relay at different line voltages are shown in Fig. 4.

Results and Discussion

Calculated friction torque at vacuum pressure condition is shown in Fig. 5. Within steady running region (rpm > 3,000) the calculated torque has a good agreement with the measured torque. Fig. 6 and Fig. 7 show the effects of valve stiffness and valve port size. According to the magnitude of gas pressure load, starting voltage required to run the compressor in steady state region changes rapidly.

Generally the electrical input of the compressor, which has ball joint type connecting rod, tends to decrease during the first few days of running because the pre-caulking torque decreases. Fig. 8 shows that electrical input decreases with running time. The starting voltage at different T_c condition and the friction torque of ball joint part at each shaft position are shown in Fig. 9, Fig. 10.

Operating time of starting relay and accelerating time to reach the intersection point of the compressor load torque curve and the motor torque curve at each voltage are shown in Fig. 4. It was found that the maximum running torque and accelerating torque were very Important in determining the starting voltage.

Conclusion

It is possible to predict the starting voltage change due to the friction change in ball joint part and the pressure condition in cylinder volume by using a computer.

A method to determine the starting voltage has been presented. According to this method, the starting voltage is determined by the accelerating characteristics, operating time of starting relay and the maximum running torque.

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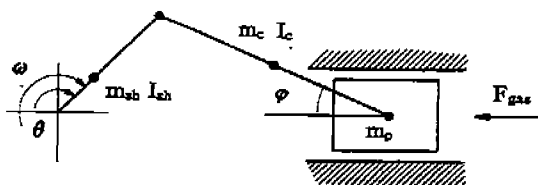


Fig. 1 Slider-crank mechanism of reciprocating compressor

Sliding parts	Coefficient of friction
shaft journal	0.17
shaft crank pin	0.18
shaft thrust	0.17
piston and cylinder	0.21
ft	0.15
fs	0.23

Table 1 Friction coefficient

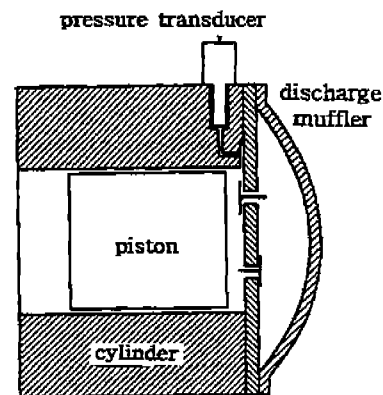


Fig. 3 Measurement of gas pressure

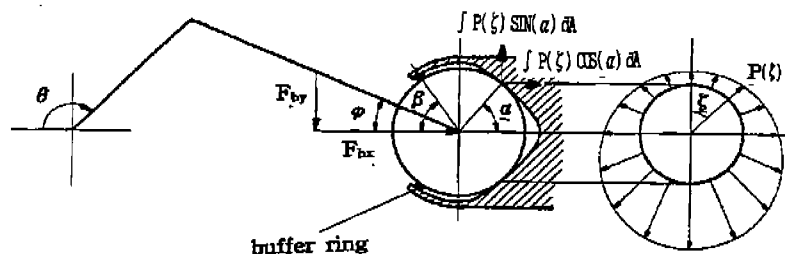


Fig. 2 Pressure distribution of spherical bearing

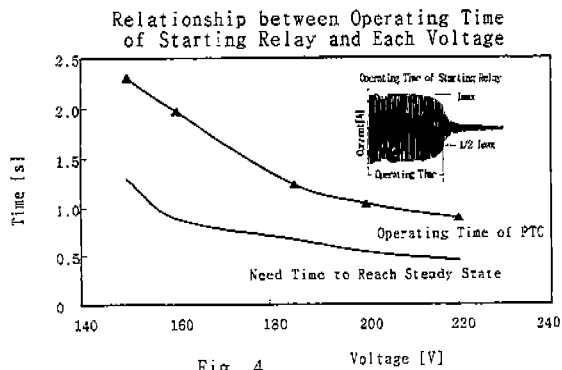


Fig. 4

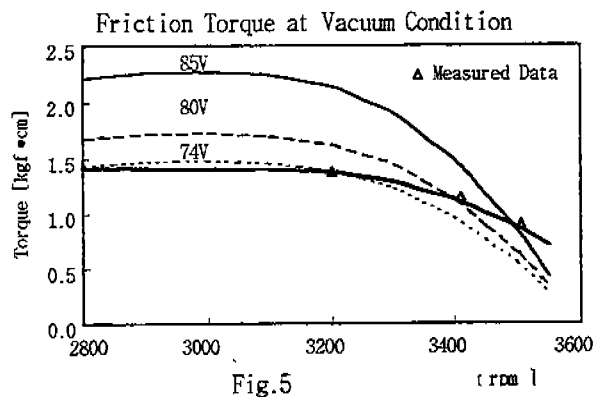


Fig. 5

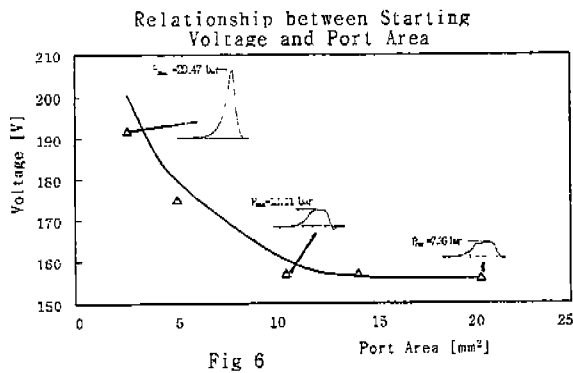


Fig. 6

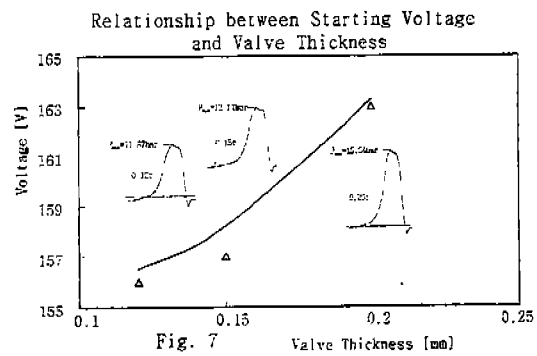


Fig. 7

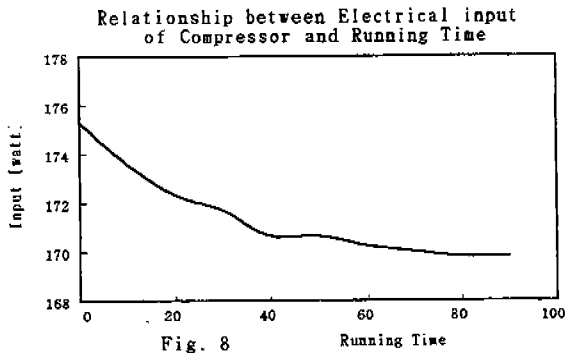


Fig. 8

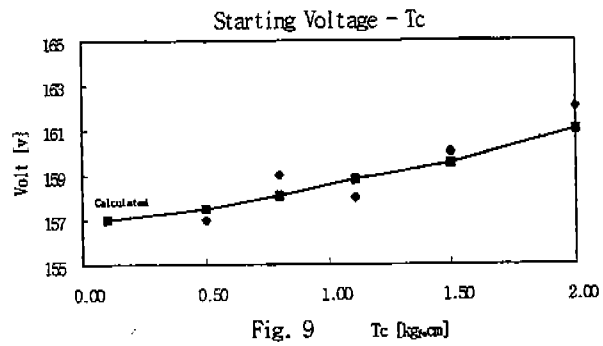


Fig. 9

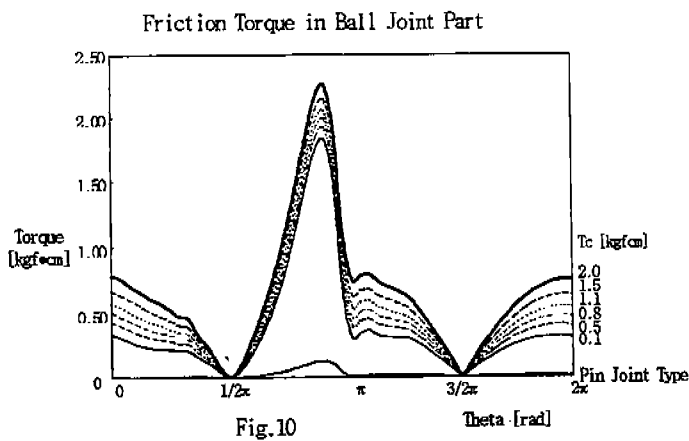


Fig. 10

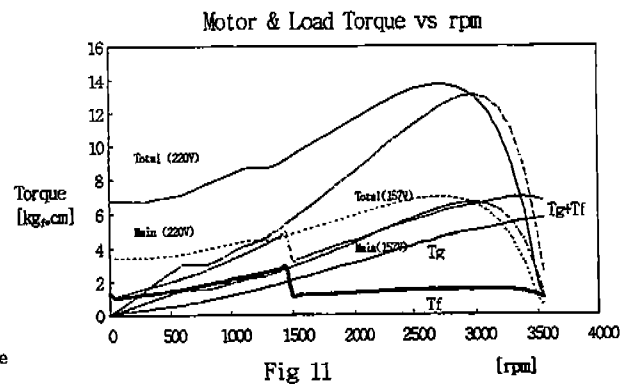


Fig. 11